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## EFFECT OF VARIABLE STATOR AREA ON PERFORMANCE OF A SINGLE-STAGE TURBINE SUITABLE FOR AIR COOLING

### IV — Stator Overall Performance With 70-Percent Design Area

by *Frank P. Behning, Bernard Bider, and Edward M. Szanca*

*Lewis Research Center*

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NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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## ABSTRACT

A program is being conducted to study a single-stage turbine at stator area settings of 70, 100, and 130 percent of design. This report presents the overall stator performance at the closed setting and compares the results with those obtained from design tests. Results are presented in terms of mass flow, outlet flow angle, and blade-surface static-pressure and velocity distributions, as well as inner- and outer-wall static pressures as obtained over a range of stator pressure ratios.

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SUMMARY

As part of a program involving the study of variable-stator turbines, a single-stage turbine is being investigated with the stator in positions opened and closed from that of design. This report presents the results of the experimental investigation of the overall performance of the stator component in the closed position. This position was set to provide a flow area at the stator outlet of 70 percent that of the design value. The stator performance presented includes that of mass flow, outlet flow angle, and blade-surface static-pressure and velocity distributions, as well as inner- and outer-wall static pressures as obtained over a range of stator pressure ratios.

The results of the investigation indicated that the mass-flow reduction at choked conditions was equal to the area reduction. The measured blade-exit flow angle and radial static-pressure distribution were close to those expected over the operating range covered. The stator operated as a convergent blade row in the closed position.

INTRODUCTION

Advanced aircraft, such as the supersonic transport, require high engine temperatures to achieve their performance goals. The turbine blading for this type of application is characterized by thick blunt profiles and low solidity because of cooling considerations. In addition to the performance requirement at design conditions, it is important in some instances that high performance be maintained at one or more off-design operational modes (e.g., ref. 1). One method considered (refs. 1 and 2) to improve off-design engine performance is the use of adjustable turbine stators. This feature would permit the turbine to operate at a fixed pressure ratio over a range of equivalent weight flow, thereby allowing the engine to operate closer to optimum cycle conditions.



One phase of the turbine research program at Lewis Research Center is the investigation of turbine performance over a range of stator areas. The turbine being used in this investigation is a 30-inch (0.762-m) cold-air turbine designed with physical features typical of those required of a turbine for high-engine-temperature application. The design and overall stator performance of the turbine are described in reference 3. The stator boundary-layer characteristics are presented in reference 4, and stage performance data are presented in reference 5. Two additional stator assemblies were fabricated to provide outlet flow areas of 70 and 130 percent of design. These were fabricated as separate units to avoid the complexities of making the stator fully variable. The investigation includes overall stator tests, stator-outlet surveys, and stage performance tests similar to those described in references 3 to 5 for the design turbine. The results obtained with the 130-percent stator have been completed and are described in references 6 to 8.

This report presents the overall results obtained with the 70-percent stator. These results include the variation in mass-flow rate, outlet flow angle, and blade-surface pressure and velocity distribution over the range of overall stator pressure ratios. These results are then compared with the corresponding results obtained from the design stator of reference 3.

## SYMBOLS

$p$	absolute pressure, lb/ft <sup>2</sup> (N/m <sup>2</sup> )
$r$	radius, in. (m)
$V$	absolute gas velocity, ft/sec (m/sec)
$w$	mass-flow rate, lb/sec (kg/sec)
$\alpha$	absolute gas flow angle measured from axial direction, deg
$\alpha_s$	blade stagger angle measured from axial direction, deg
$\delta$	ratio of inlet pressure to U. S. standard sea-level pressure
$\theta_{cr}$	squared ratio of critical velocity at turbine inlet to critical velocity of U. S. standard sea-level air

### Subscripts:

$cr$	condition at Mach 1
$d$	station downstream of turbine stator
$h$	hub (inner wall)

- i station at stator-blade-exit orthogonal
- t tip (outer wall)
- 0 station at turbine inlet
- 1 station at stator-outlet freestream

Superscript:

- (') total-state condition

## STATOR MODIFICATION

The setting of the closed stator was determined by decreasing the mean-section channel-exit orthogonal length to a value 30 percent less than that of the design stator. Figure 1 presents the blade positions at the mean section for both stators. The area change was effected by rotating the blade about the centerline of the trailing-edge radius, thereby maintaining the trailing edge at the same axial position. The blade stagger angle increased from  $41.03^\circ$  to  $48.82^\circ$ , a change of  $7.79^\circ$ . This change in blade setting resulted in a decrease of the channel-exit orthogonals at the hub and tip stations of 35 and 26 percent, respectively.

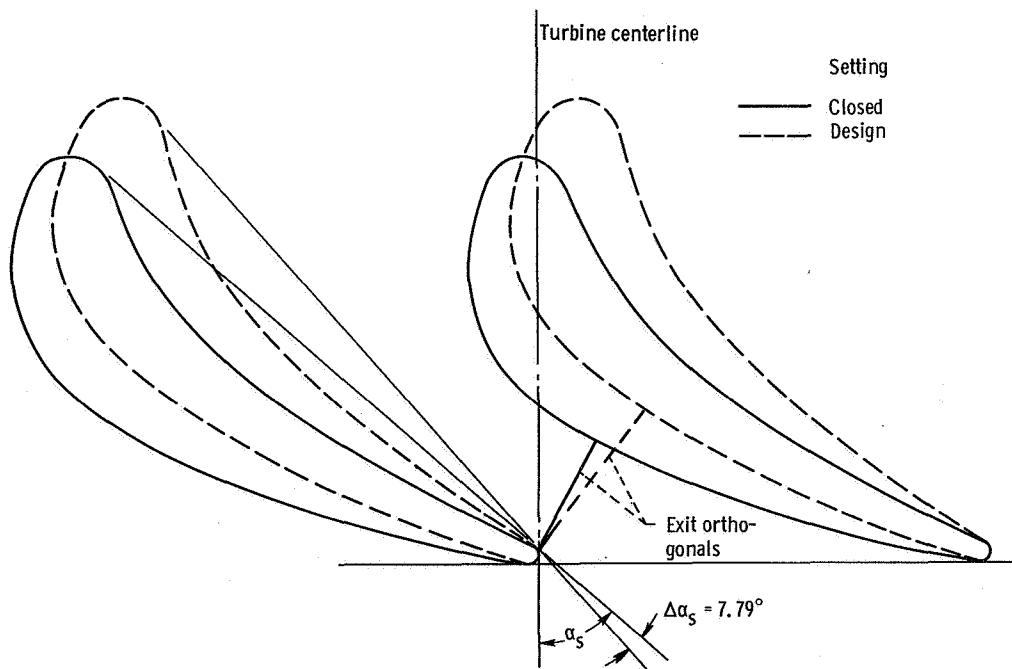


Figure 1. - Relative blade positions of closed and design-area stators.



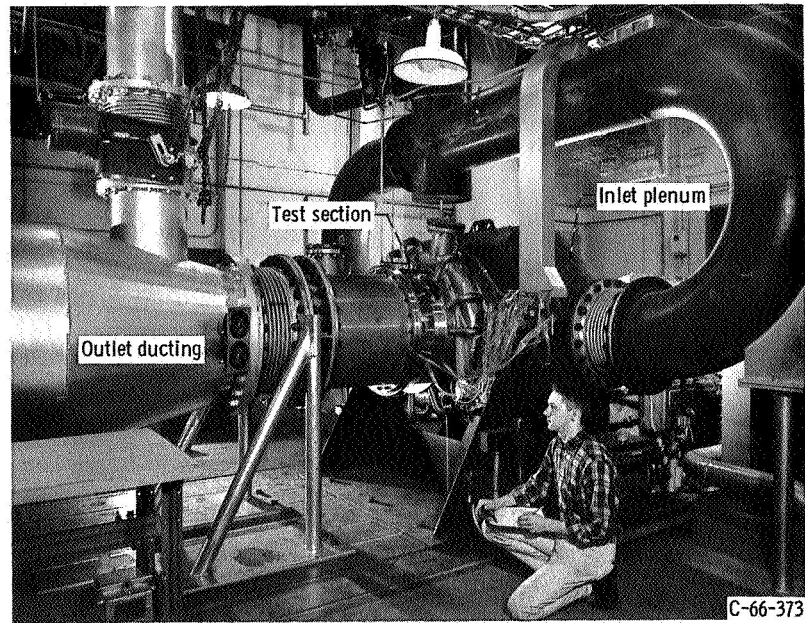


Figure 2. - Test facility.

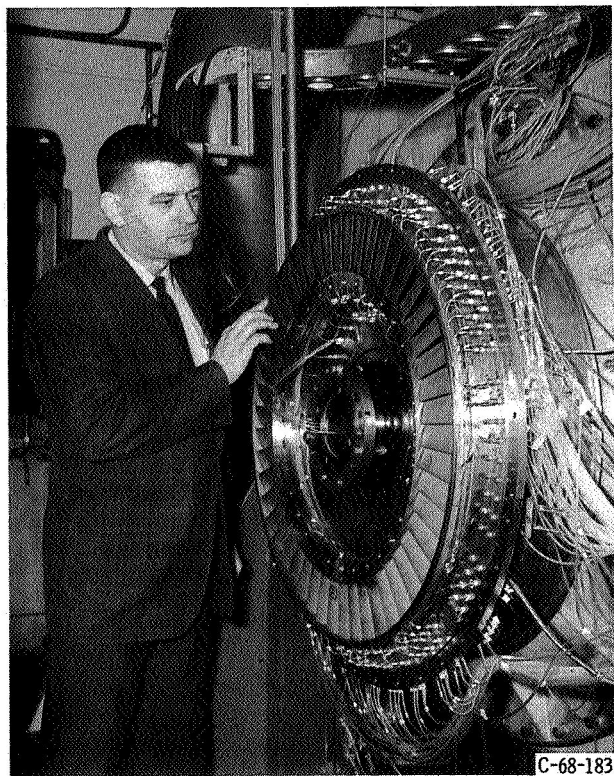


Figure 3. - Closed-stator assembly installed in test facility.

The increase in outlet flow angle  $\alpha_1$ , resulting from the stagger-angle  $\alpha_s$  change, was calculated by using the method of reference 3 and amounted to  $7.2^\circ$  at the mean section.

## APPARATUS, INSTRUMENTATION, AND PROCEDURE

The test facility used in this program was designed to incorporate a complete turbine. The test facility is shown in figure 2. The stator was tested as a separate component for this phase of the program. The closed stator installed in the facility is shown in figure 3.

### Apparatus

The apparatus consisted of the closed-stator assembly and test sections, together with suitable ducting and control valves. A diagrammatic sketch of the turbine-stator test section is shown in figure 4. The stator had a 30-inch (0.762-m) outside diameter. It is a full annular cascade comprised of 50 hollow, cast, stainless-steel blades. The blade height was 4 inches (0.1016 m). As mentioned in the STATOR MODIFICATION section, the blades were oriented so as to provide a mean-section channel-exit orthogonal length 30 percent less than that of the design stator.

Dry pressurized air was supplied to the test section from the combustion air system. The air flow was measured with a calibrated Dall tube, which is a specialized type of venturi meter. Turbine-inlet pressure was controlled by butterfly throttle valves located between the Dall tube and the turbine-inlet plenum chamber. Air leaving the stator was ducted to the exhaust system. The stator-outlet pressure was controlled by butterfly throttle valves located downstream of the test section.

### Instrumentation

The stator test section was instrumented similarly to the design turbine-stator test section (ref. 3). At each of the four measuring stations (fig. 4) were located a total of eight static-pressure taps, four each at the inner and outer walls. The inner- and outer-wall taps were located opposite to each other. At the stator-inlet and downstream measuring stations, 0 and d, respectively, the pressure taps were spaced about  $90^\circ$  apart around the circumference. The pressure taps at station i were situated at the center of the exit orthogonal of four stator passages spaced about  $90^\circ$  apart. At station 1 the pressure taps were located on the projected passage centerline of the same four passages.



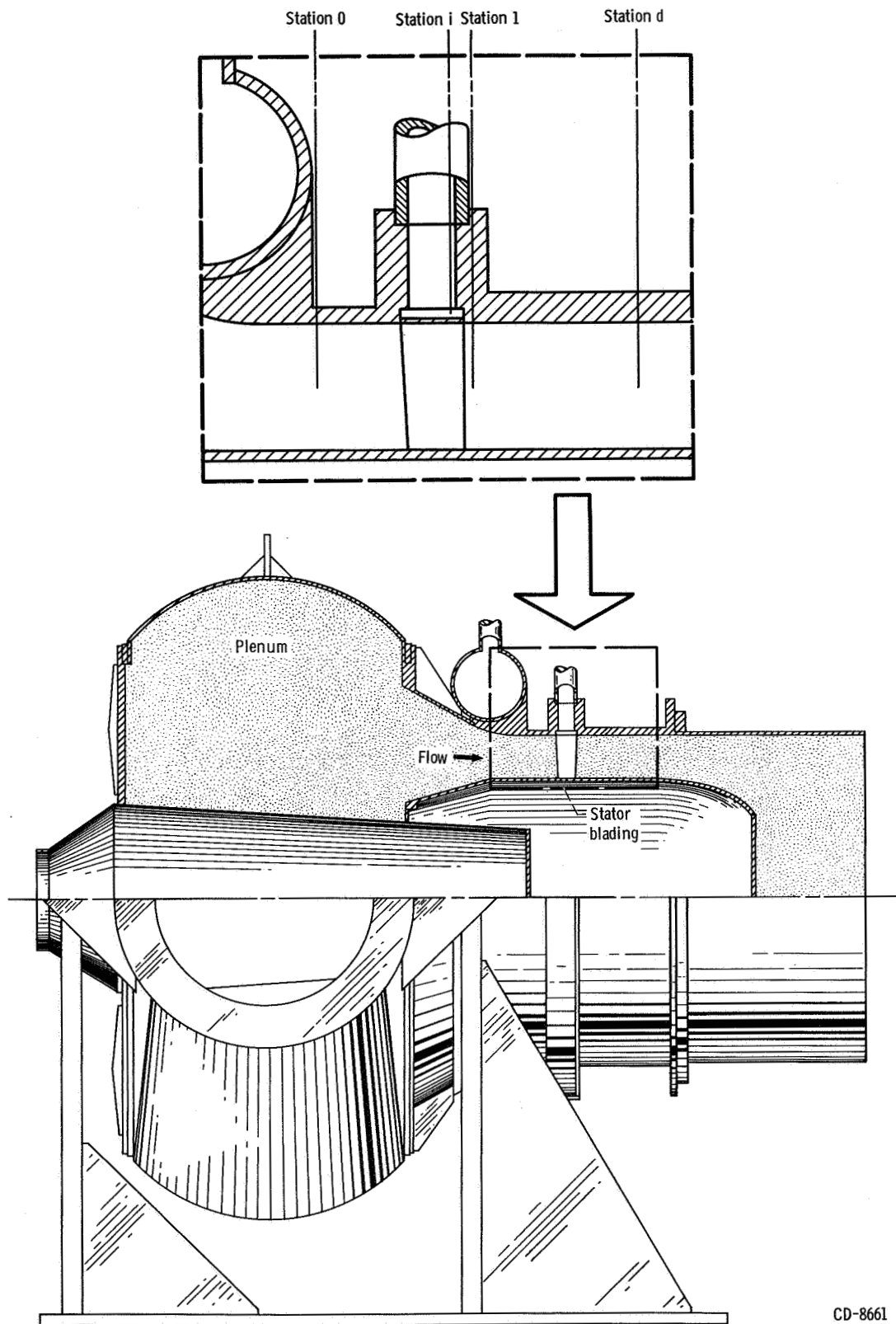


Figure 4. - Schematic diagram of turbine-stator test section.

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Two thermocouple rakes were located at the stator-inlet measuring station spaced  $180^\circ$  apart circumferentially. Each rake contained five thermocouples situated at the area center radii of five equal annular areas.

Four Kiel-type total-pressure probes were also located at the inlet to the stator. The probes were positioned at the area center radius and were spaced  $90^\circ$  apart circumferentially.

Each station group of four pressure taps, except those at station d, were manifolded and connected to a single manometer tube. The eight outlet static pressures at station d were individually read on manometer tubes.

A total of 78 static-pressure taps were installed on the blade suction and pressure surfaces at the hub, mean, and tip sections of three blade passages. The static-pressure taps at the hub and tip sections were located 0.1 inch (0.254 cm) from the inner and outer walls to avoid end wall boundary layers. The mean-section pressure taps were located midway between the inner and outer walls. The locations of the blade static taps are shown in figure 5 for the mean section. The static-pressure taps were distributed approximately equally among the three blade sections.

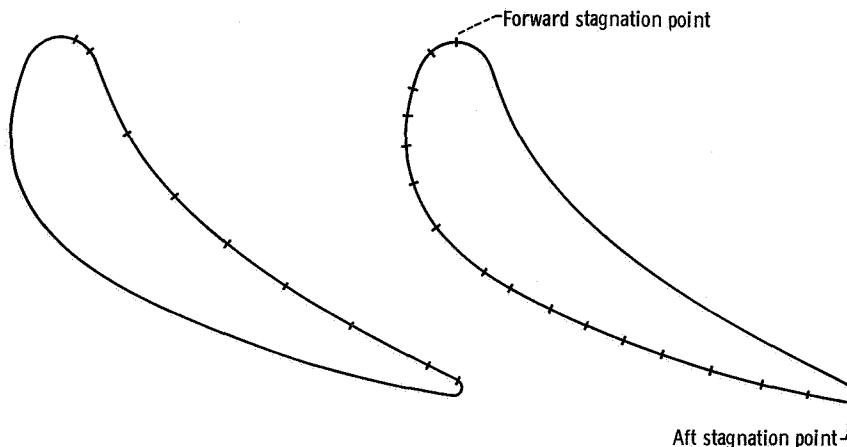


Figure 5. - Mean blade section showing location of blade-surface static-pressure taps.

All pressures were read on mercury fluid manometers except for the Dall-tube differential pressure which was read on a tetrabromoethane fluid manometer. All temperatures were read with a direct-reading self-balancing potentiometer.

The mass-flow rate was measured with the Dall tube. The flow angle at the stator exit (station 1) was measured with a self-aligning angle probe and indicated on an X-Y plotter.



## Procedure

The test procedure for the closed stator was identical to the test procedure for the design stator of reference 3. The total pressure at the stator inlet was set at a nominal value of 30 inches of mercury absolute ( $1.0159 \times 10^5 \text{ N/m}^2 \text{ abs}$ ). The total inlet temperature was approximately  $530^\circ \text{ R}$  ( $294.3 \text{ K}$ ). The mass-flow data were obtained over a range of overall pressure ratios by varying the outlet pressure.

The static pressures along the blade surfaces and the stator-exit flow angle data were obtained at pressure ratios corresponding to nominal outlet-hub critical-velocity ratios  $(V/V_{cr})_{d,h}$  of 0.38, 0.5, 0.7, 0.896 (reference design), 1.1, and 1.3.

The outlet flow angle  $\alpha_1$  of the gas stream was measured at the stator exit. This angle measurement was recorded on a X-Y plotter as a function of circumferential location for various radial positions and over the range of critical-velocity ratios  $(V/V_{cr})_{d,h}$ .

## RESULTS AND DISCUSSION

The results of the subject investigation are presented in four parts: (1) mass-flow and radial static-pressure distribution, (2) stator-exit-angle variation, (3) blade-surface static pressure and velocity distribution as obtained at the hub, mean, and tip sections, and finally (4) hub and tip wall static pressures as obtained at the channel-exit orthogonal. Where appropriate, comparison is made with results obtained from the design stator.

### Mass Flow

The experimentally obtained equivalent mass flows  $w\sqrt{\theta_{cr}}/\delta$  are shown in figure 6 together with that obtained for the design stator setting (ref. 3). Any given data point appears as two points since, for a given weight flow, there is a resulting pressure ratio at the inner wall  $p'_0/p_{d,h}$  and one for the outer wall  $p'_0/p_{d,t}$ . Also shown are the predicted trends of wall static pressures as a function of weight flow calculated by the method described for the design stator in reference 3.

The figure indicates that the maximum flow passed by the closed stator was 30.0 pounds per second (13.6 kg/sec), which is 30 percent less than the maximum value for the design stator. At the design hub pressure ratio for the design stator of 1.705, the predicted flow for the closed stator is noted to be 28.2 pounds per second (12.8 kg/sec), a decrease of 29.3 percent from the design stator. The obtained flow at this condition is seen to be 28.6 pounds per second (13.0 kg/sec), which is about 1 percent higher than that predicted. Figure 6 also shows that the radial pressure gradients obtained for the

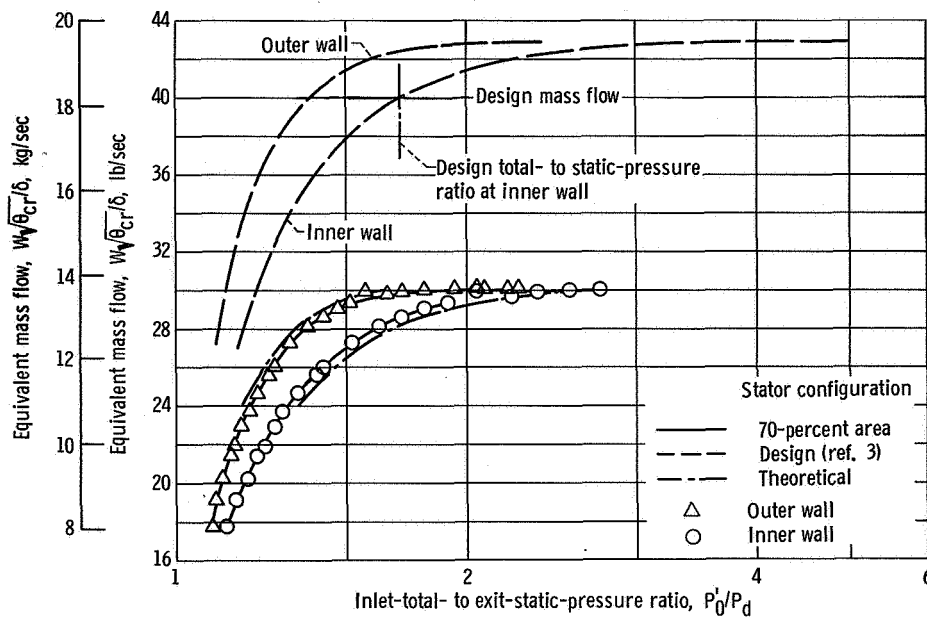


Figure 6. - Variation of equivalent mass flow with total- to static-pressure ratio at inner and outer walls.

closed stator are somewhat less than those predicted. Finally, it is noted that closing the stator caused choking to occur at a lower overall pressure ratio.

## Outlet Flow Angle

The experimental stator outlet flow angle  $\alpha_1$  is shown in figure 7 as a function of radius at the hub critical-velocity ratio  $V/V_{cr}$  of 0.896. Also shown is the variation in theoretical angle for the subject stator. The circumferential variation in this angle mea-

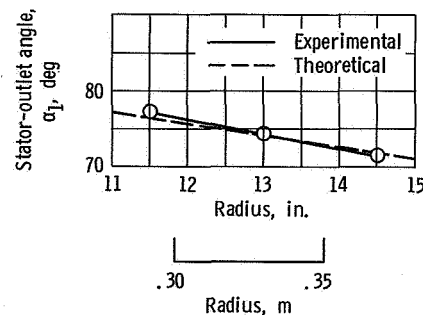


Figure 7. - Comparison of experimental stator-outlet flow angle with theoretical for critical-velocity ratio of 0.896.



surement was  $\pm 2\frac{1}{2}^{\circ}$  in the region outside the wake areas and was averaged to obtain the test points shown. At the mean radius, the experimental outlet angle from figure 7 is  $74.35^{\circ}$  which is close to the predicted value of  $74.2^{\circ}$ .

The outlet flow angle was also found to be essentially insensitive to Mach number level. At any given radius, the measured angle varied less than  $\pm 0.25^{\circ}$  over the range of velocities investigated.

## Blade-Surface Static-Pressure and Velocity Distributions

The surface static-pressure distributions along the suction and pressure surfaces at the hub, mean, and tip sections of the stator passage are shown in figure 8. These pressure distributions were obtained for a range of hub critical-velocity ratios from 0.38 to 1.3, inclusively. The curves show the ratio of the surface static pressure to the inlet total pressure plotted as a function of nondimensional blade-surface length. Values of 0 and 1 correspond to the apparent forward and aft stagnation points. These pressure-tap locations are shown in figure 5 for the mean section and are typical of the hub and tip sections.

The pressure decreases, generally, with increasing critical-velocity ratio. On the suction surfaces (figs. 8(b), (d), and (f)) there is less expansion downstream of the closed channel at the higher critical-velocity ratios than for the design stator. It also appears from the figure that choking occurs first at the hub region and progresses radially outward.

The static-pressure data corresponding to  $(V/V_{cr})_{h,d}$  of 0.896 were converted into surface critical-velocity ratios by assuming isentropic free-stream conditions as in reference 3. Figure 9 presents the resulting velocity distribution along with the theoretical velocity distribution that was obtained by using the experimental value of weight flow and the method of reference 9. Isentropic free-stream conditions were also assumed in this procedure.

It is seen from figure 9 that the theoretical curves for the closed stators approximated the trends obtained from the experimental data. This comparison also shows that the loading in the forward part of the blade (as indicated by the difference between suction and pressure-surface velocities) is not as high as that predicted by the design procedure. The probable explanation for this difference is that, because of the low solidity, the stream-filament theory which is used in reference 9 is less adaptable to this portion of the channel.

The experimental velocity distribution for the closed blade is similar to that for the design blade. The peak tip suction-surface velocity is less for the closed stator, which results in a smaller suction-surface diffusion.

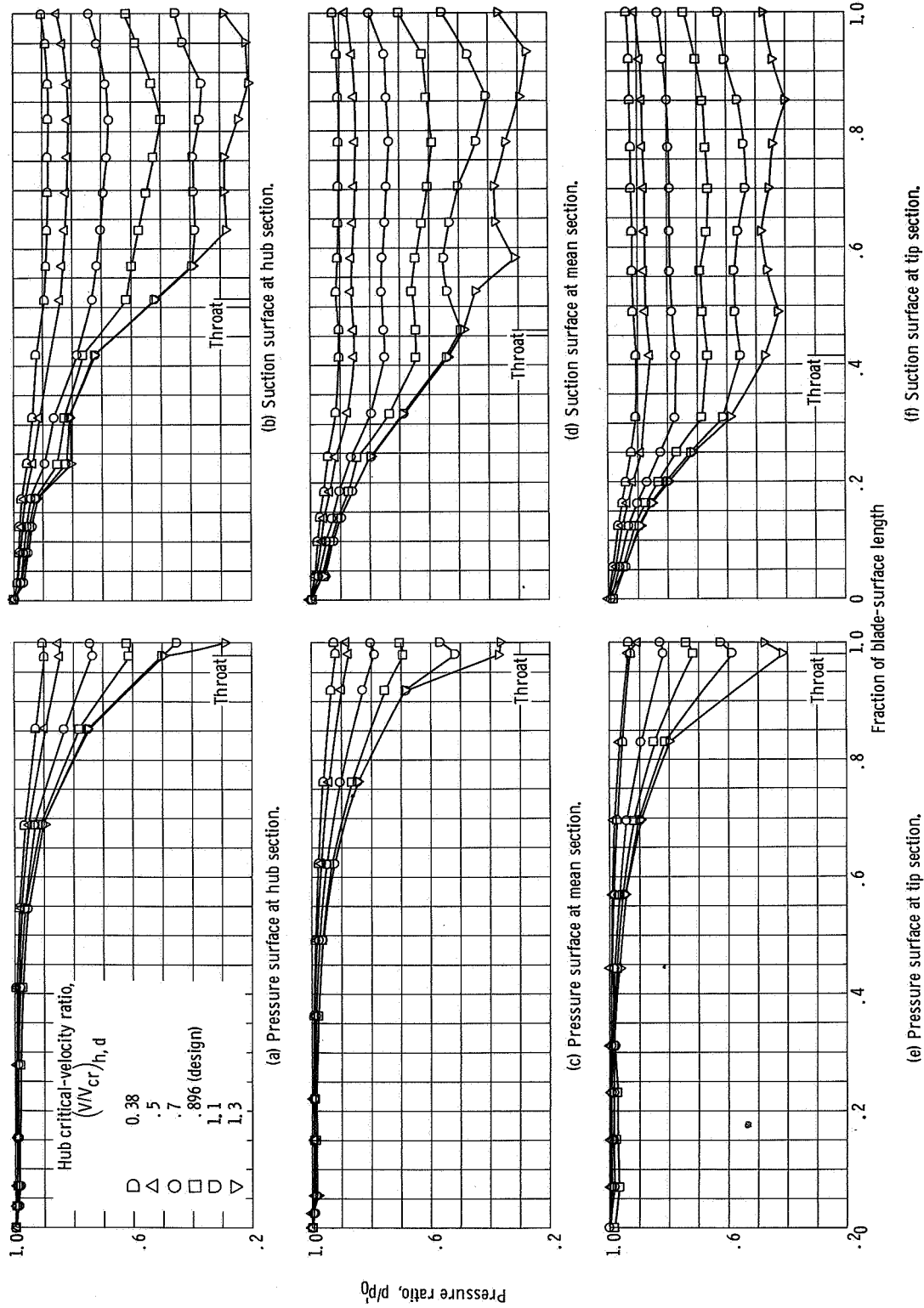


Figure 8. - Blade-surface pressure distribution.

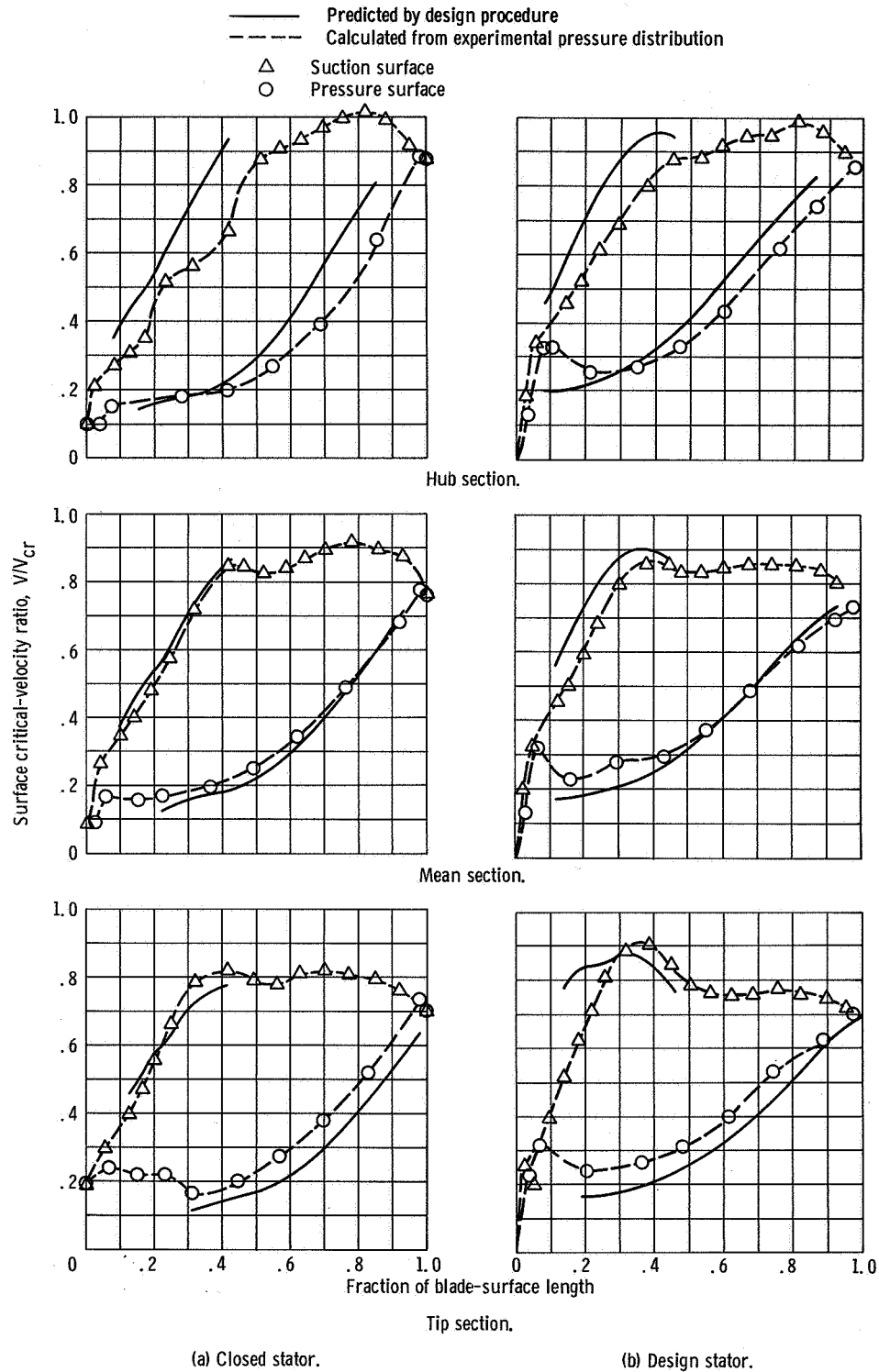


Figure 9. - Comparison of predicted surface velocity with experimentally obtained surface velocity at design pressure ratio for closed and design stators.



## Channel-Exit Pressure Ratios

In the APPARATUS, INSTRUMENTATION, AND PROCEDURE SECTION, it was indicated that static-pressure taps were located on the center of the exit orthogonal at the hub and tip. These taps are assumed to be at the throat position and are used to study the stator choking conditions.

The experimentally obtained pressure ratios at the throat (station i) are shown in figure 10. From this figure it is seen that both the hub and tip sections act as a conver-

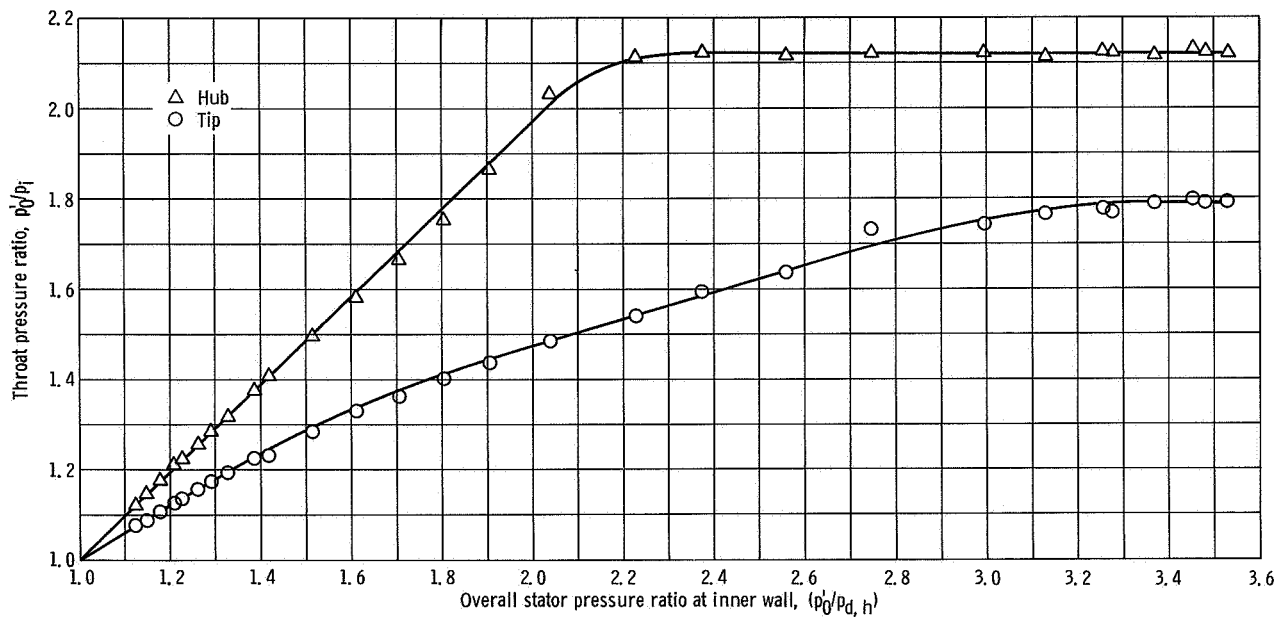


Figure 10. - Variation of stator throat pressure with overall stator pressure ratio for closed stator.

gent section. This is indicated by the trend of throat static pressures to approach constant values with increasing overall pressure ratio. The throat pressures become constant at values of  $p'_0/p_{d,h}$  of about 2.4 for the hub and about 3.3 for the tip.

## SUMMARY OF RESULTS

As part of a single-stage-turbine, variable-stator program, an investigation of the overall performance of the stator in a closed position was made. In this configuration the channel-exit orthogonal at the mean section was decreased to 70 percent that of the design setting. The results of the investigation can be summarized as follows:

1. In its closed position, the stator mass flow at choked conditions was proportional

to the corresponding area change.

2. The measured flow angle and radial pressure distribution at the blade exit were close to that expected over the operating range covered. The blade-exit flow angle was also found to be essentially insensitive to Mach number level.

3. A comparison of the blade-surface velocities obtained experimentally with those evolved from the design procedure showed that the design procedure predicted fairly well the shape and magnitude of the blade-surface velocities.

4. In its closed position, the stator operated as a convergent blade row.

Lewis Research Center,  
National Aeronautics and Space Administration,  
Cleveland, Ohio, July 19, 1968,  
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